COMPARISON OF SIMULATED AND MEASURED TEST DATA ON AIR-SOURCE HEAT PUMPS

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KEY WORDS

Air-source heat pumps, simulation, measured, test.

ABSTRACT

Simulations were performed on four sets of manufacturer supplied heat pump data to assess the ability of an air-source heat pump computer simulation model to predict operating performance. Each set of manufacturer data consisted of a minimum efficiency (10 SEER) and high-efficiency (12 to 13 SEER) single-speed heat pump model. Detailed design data were provided for each heat pump model in addition to test data measured at each of the four steady-state test conditions in the United States' Department of Energy's (DOE) air-source heat pump test procedure. Simulation results were compared to the measured test data for three of the four steady-state test conditions (one cooling and two heating). By incorporating manufacturer suggested modeling changes and simulation practices, average differences between simulated results and test data for capacity, efficiency, and compressor power were either under or just above 5% at the DOE "A" cooling test condition (outdoor ambient air temperature of 95°F (35°C)). Average differences between simulated results and test data at both the high-temperature (outdoor ambient air temperature of 47°F (8.3°C)) and low-temperature (outdoor ambient air temperature of 17°F (-8.3°C)) heating test conditions were not as close. As a result, further model modifications were made yielding simulation results which were within 5% of the measured capacity, efficiency, and compressor power at the high-temperature heating test condition. Average differences between simulation results and test data at the low-temperature heating test condition were still beyond 5% but much improved over the average simulation vs. test data differences resulting from the first set of model modifications.

INTRODUCTION

Air-source heat pumps and central air conditioners with cooling capacities rated below 19 kW (65,000 Btu/hr) are one of the thirteen appliances regulated for efficiency by the United States' National Appliance Energy Conservation Act (NAECA) [1]. For the appliances regulated by NAECA, the Act requires periodic updates to the minimum efficiency standards. As a result, an engineering analysis was initiated in 1992 to determine, among other things, the efficiency increases possible through changes in heat pump and air conditioner design. A computer simulation model was used to estimate the efficiency changes associated with specific equipment modifications. In order to ensure that the simulation model provided reasonable results, air-source heat pump and central air conditioner manufacturers provided design and measured test data to validate the performance of the model. In order to facilitate the exchange of data, several meetings were held between representatives from air-source heat pump and central air conditioner manufacturers, the Air-Conditioning and Refrigeration Institute (ARI) (trade association for the air-conditioning industry), Lawrence Berkeley National Laboratory (LBNL), the U.S. Department of Energy (DOE), and Oak Ridge National Laboratory (ORNL). Six meetings were held spanning a time period from January 1992 to March 1993 with the primary objective of these discussions being the review of the computer simulation model. Upon review of several simulation results, manufacturers provided input on how to improve the performance of the simulation model as well as suggesting guidelines in which to conduct simulations.

SIMULATION MODEL

The Oak Ridge National Laboratory Modulating Heat Pump Design Tool [2] was the computer simulation model used for the engineering analysis. This simulation model consists of a modulating heat pump design model and a parametric-analysis (contour-data generating) front-end. Collectively the simulation model is also referred to as MODCON which is in reference to its modulating and contour data generating capabilities.

MODCON is an extension of the Oak Ridge National Laboratory Mark III Heat Pump Design Model [3] [4]. The Mark III model is a comprehensive program for the simulation of single-speed, electrically driven, vapor compression, air-source heat pumps and air conditioners. It is a steady-state model that is able to calculate the energy efficiency of the equipment being modeled at specified ambient conditions. The Mark III model is divided into two main parts; the high side and the low side. The high side includes models for the compressor, the condenser, and the expansion device, while the low side contains the evaporator. The model first performs a high-side balance based on calculating a mass flow rate through the flow control device that matches the one determined for the compressor. Once a high-side balance is achieved, a low-side balance is performed in which the evaporator model seeks an air inlet temperature that ensures the previous balance at the high side.

The primary capability distinguishing MODCON from its predecessor is its ability to predict the steady-state performance of two-speed and variable-speed systems. In addition to its ability to model variable-speed compressors and fans, MODCON also includes the following capabilities and improvements not found in the Mark III version: 1) substantially improved and extended air-side heat exchanger correlations for modulating applications, 2) a refrigerant charge inventory option allowing the user to either specify or determine the required charge, 3) a provision for variable-opening flow controls used in modulating heat pumps (e.g. electronic and thermostatic expansion valves), 4) a provision for input selection of refrigerant, and 5) an automated means to conduct parametric performance mapping of selected pairs of independent design variables.

Two changes were made to MODCON before any attempt was made to validate its capability to estimate the performance of central air-conditioning and heat pump systems. The first change was to incorporate new compressor map-fitting routines based on ARI standard 540-91 (Method for Presentation of Compressor Performance Data) [5]. By making this change, MODCON uses the same convention that is used by the heat pump and air-conditioning industry to represent compressor performance. Prior to this change, MODCON used bi-quadratic functions for both the input power and mass flow rate as a function of evaporating and condensing temperatures to describe the compressor. The ARI standard is based on the use of a third order polynomial equation which is provided below.

$$f(T_{inlet}, T_{outlet}) = C_1 + C_2 T_{inlet} + C_3 T_{outlet} + C_4 T_{inlet}^2 + C_5 T_{inlet} T_{outlet} + C_6 T_{outlet}^2 + C_7 T_{inlet}^3 + C_8 T_{outlet} T_{inlet}^2 + C_9 T_{inlet} T_{outlet}^2 + C_{10} T_{outlet}^3 \tag{1}$$

where T_{outlet} = condenser saturation temperature
T_{inlet} = evaporator saturation temperature

The constants (C_1 through C_{10}) are derived through the use of a map-based compressor model and are used as inputs to MODCON to describe the compressor. Detailed descriptions of the map-based compressor model are provided in ARI standard 540-91.

The second change added the capability to MODCON of iterating on a specified system capacity. This change was based on manufacturer recommendations to conduct the engineering analysis by holding system capacity constant when analyzing design option improvements. For most of the design options analyzed, increases in system capacity result when design improvements are made. In order to hold system capacity constant, MODCON was modified to vary the compressor displacement to meet the specified system capacity. Once the compressor displacement is varied, the user must resize the compressor motor to ensure that the efficiency of the motor is identical to that of the baseline ("non-altered") compressor. Resizing of the motor is based on the percentage of the baseline compressor's nominal torque that is required at the new compressor displacement. An iterative loop was added to MODCON to allow for iteration on system capacity. New input variables for the specified system capacity and the acceptable tolerance on capacity (for purposes of iterative convergence) were added to MODCON.

VALIDATION OF SIMULATION MODEL

In order to validate the performance of the MODCON simulation model, actual heat pump models had to be simulated. The validity of the simulation results then had to be compared to test data. Thus, in addition to providing design data describing in detail the physical characteristics of heat pump models, manufacturers also provided accompanying test data at each of the steady-state rating conditions specified in the U.S. Department of Energy (DOE) test procedure. Table 1 summarizes the operating conditions for the various steady-state rating conditions which single-speed central heat pumps must be tested to. As cited by the U.S. DOE test procedure [6], these operating conditions are found in ARI Standard 240-77 [7] for unitary air-source unitary heat pump equipment.

Table 1: Operating Conditions for DOE Single-Speed Rating Conditions

	Letter Designation for U.S. DOE test	**Indoor Unit - Air Entering**				**Outdoor Unit - Air Entering**			
		Dry Bulb		Wet Bulb		Dry Bulb		Wet Bulb	
U.S. DOE Test Description		*°C*	*°F*	*°C*	*°F*	*°C*	*°F*	*°C*	*°F*
"A" Cooling Steady State	A	26.7	80	19.4	67	35	95	23.9	75
"B" Cooling Steady State	B	26.7	80	19.4	67	27.8	82	18.3	65
High-Temp. Heating Steady	E	21.1	70	15.6	60	8.3		47	6.143
Low-Temp. Heating Steady	H	21.1	70	15.6	60	-8.3		17	-9.415

Engineering and test data were provided through the manufacturers' trade organization, the Air-Conditioning and Refrigeration Institute (ARI). ARI gave letter designations to each manufacturer providing data. The four manufacturers which test versus simulation comparisons were conducted were designated by ARI as Manufacturers A, B, D, and J. For each of these four manufacturers, simulations were conducted at the operating conditions specified in Table 1 on two heat pump models (a minimum- and high-efficiency single-speed model). For the remainder of the report, all DOE test procedure conditions will be referred to by their corresponding outdoor dry bulb temperature in degrees Fahrenheit (e.g., the DOE "A" test condition will be referred to as the 95°F (35°C) test condition).

The design data provided by the four manufacturers provided the necessary information for MODCON to describe a particular unit's physical characteristics. To carry out the simulations, measured quantities from the test data were also required by MODCON and were used to specify the following input variables: 1) the amount of superheat, 2) the amount of subcooling, 3) the air flow rate over the outdoor heat exchanger coil, 4) the outdoor fan motor power consumption, 5) the air flow rate over the indoor heat exchanger coil, 6) the indoor fan motor power consumption, 7) the compressor shell heat loss rate, 8) the refrigerant temperature rise in the suction line, 8) the refrigerant pressure drop in the suction line, 9) the refrigerant pressure drop in the outdoor heat exchanger coil, and 10) the refrigerant pressure drop in the indoor heat exchanger coil. An input variable to MODCON called the refrigerant-side pressure drop multiplier was used to match simulated heat exchanger pressure drops with those provided in the test data.

Comparison of Simulation Results to Manufacturer Test Data

Figures 1 through 4 depict the percentage differences between the test data and the simulation results for capacity, efficiency (expressed as the energy-efficiency ratio or EER[1]), and compressor power. There is one figure per manufacturer with each figure divided into two sections; the section on the left details the comparisons for the minimum efficiency model (nominal cooling efficiency of 10 SEER[2]) while the section on the right details the comparisons for the high-efficiency model (nominal cooling efficiency of either 12 or 13 SEER). Comparisons for the 95°F (35°C), 47°F (8.3°C), and 17°F (-8.3°C) test conditions are provided (results for the 82°F (27.8°C) test condition are not provided as they are extremely similar to those for the 95°F (35°C) test condition). In each figure, labels are under each set of comparisons for capacity, efficiency, and compressor power designating the corresponding model and test condition (e.g., 10 SEER-95°F designates data comparisons for the 10 SEER model rated at the 95°F (35°C) test condition).

To establish an acceptable level of error in the simulation results, allowable tolerances on reporting compressor performance data are used as a guideline. According to ARI Standard 540-91 on presenting compressor performance data, an allowance of 5% due to testing and manufacturing variations is allowed on the standard ratings of compressors (the allowance for capacity cannot be lower than -5% while the allowance for power input can be no more than +5%) [8]. Based on these acceptable tolerances on reporting compressor performance data, the simulation results provided in Figures 1 through 4 are judged to be accurate if they are within ±5% of the test data.

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[1] The U.S. DOE test procedure defines the EER as the cooling capacity in Btu/hr divided by the power input in watts. Although the EER is used in all subsequent Figures to describe heat pump heating efficiency comparisons, the efficiency descriptor for heat pumps is the COP. Internationally, the COP is defined as the heating capacity in watts divided by the power input in watts.

[2] The efficiency of the heat pump models are expressed in terms of their cooling efficiency; the SEER or seasonal energy efficiency ratio. It is customary in the U.S. for manufacturers to define the efficiency of a heat pump by its SEER. The U.S. DOE test procedure defines the SEER as the total cooling in Btu's during its normal annual usage period for cooling divided by the total electrical energy use in watt-hours during the same period.

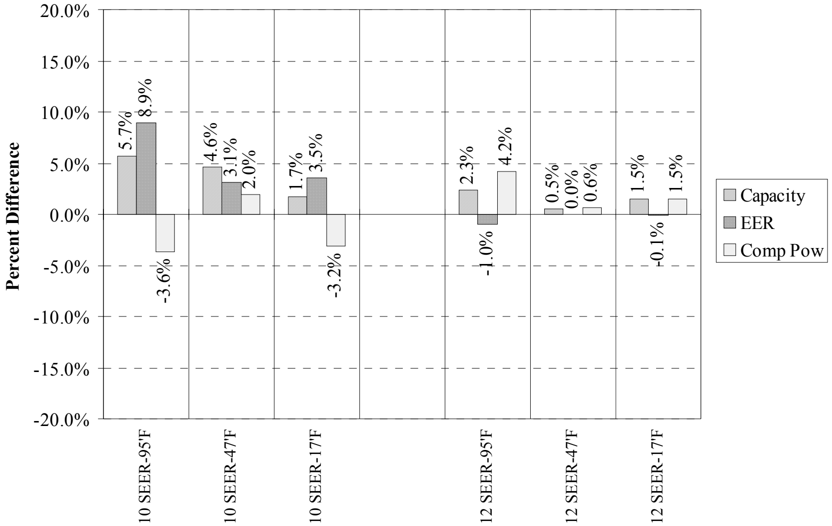

Figure 1: Manufacturer A, Test Data vs. Simulation Results

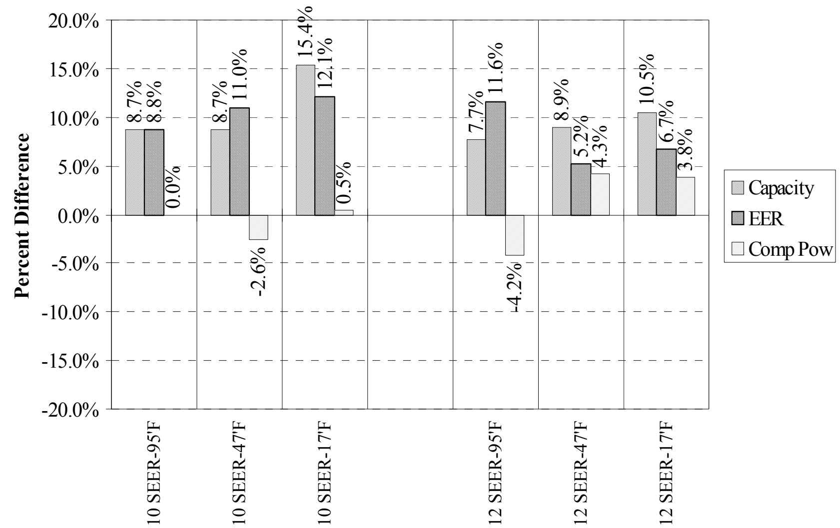

Figure 2: Manufacturer B, Test Data vs. Simulation Results

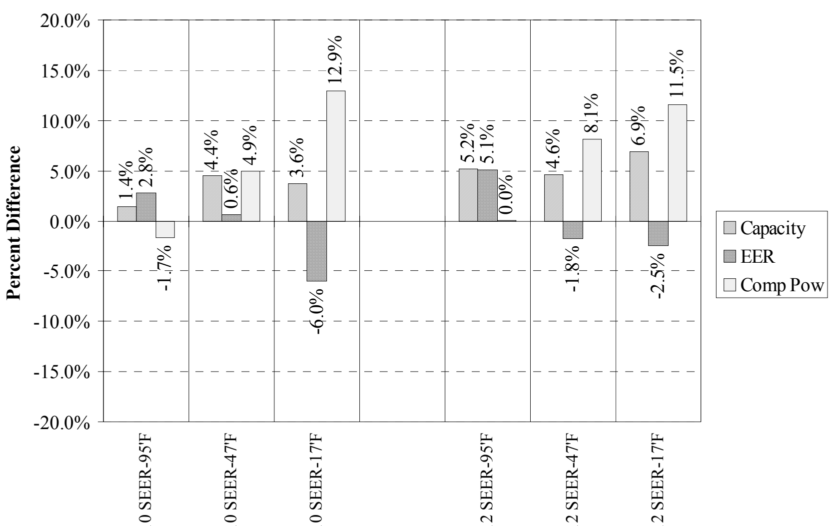

Figure 3: Manufacturer D, Test Data vs. Simulation Results

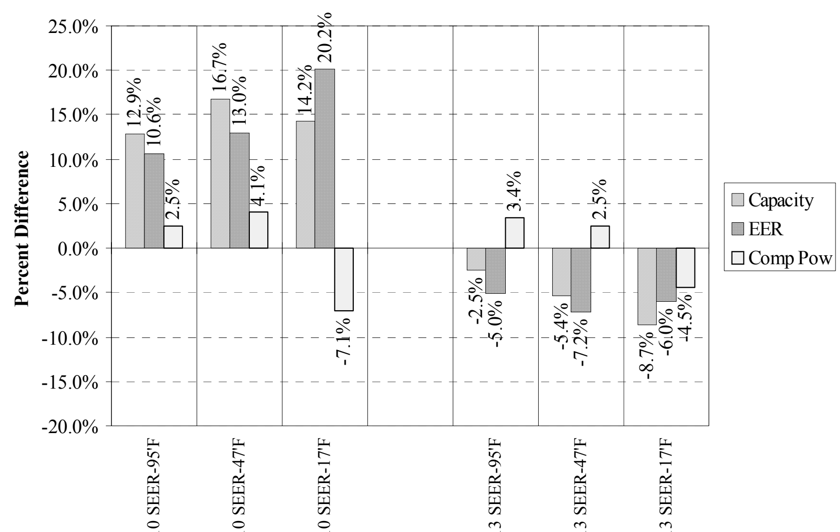

Figure 4: Manufacturer J, Test Data vs. Simulation Results

With regard to the data comparisons being made, a significant portion are beyond the 5% tolerance level. Starting with the heat pump's cooling side, of the 24 comparisons being depicted between simulated and measured values in Figures 1 through 4, 42% (or ten simulated quantities) are beyond the 5% tolerance level. But, differences of over 10% between simulated and measured values occur only for a couple of systems; the capacity and efficiency for Manufacturer J's 10 SEER system and the efficiency for Manufacturer B's 12 SEER system. Average differences for capacity, efficiency, and compressor power for all four sets of manufacturer data are 5.8%, 6.7%, and 2.4%, respectively. On an average basis, MODCON is doing an adequate job at predicting heat pump cooling performance. The average differences for capacity and efficiency do exceed the 5% tolerance level, but only by amounts of 0.8 and 1.7 percentage points, respectively.

With regard to the heat pump's heating side, separate data analyses had to be performed for the 47°F (8.3°C) and the 17°F (-8.3°C) test conditions. The outdoor ambient temperature conditions are great enough to warrant separate analyses. At the 47°F (8.3°C) test condition, of the 24 comparisons depicted between simulated and measured values in Figures 1 through 4, 38% (or nine simulated quantities) are beyond the 5% tolerance level. But differences of over 10% between simulated and measured values occur only for a couple of systems; the capacity and efficiency for Manufacturer J's 10 SEER system and the efficiency for Manufacturer B's 10 SEER system. Average differences for capacity, efficiency, and compressor power for all four sets of manufacturer data are 6.7%, 5.2%, and 3.6%, respectively. On an average basis, MODCON is doing an adequate job at predicting heat pump heating performance at the 47°F (8.3°C) test condition. The average differences for capacity and efficiency do exceed the 5% tolerance level, but only by amounts of 1.7 and 0.2 percentage points, respectively.

At the 17°F (-8.3°C) test condition, average differences between simulated and measured capacity, efficiency, and compressor power for all four sets of manufacturer data are greater than those at the 47°F (8.3°C) test condition. Average differences are 7.8% for capacity, 7.1% for efficiency, and 5.6% for compressor power at the 17°F (-8.3°C) test condition. Of the 24 comparisons depicted between simulated and measured values in Figures 1 through 4, 54% (or thirteen simulated quantities) are beyond the 5% tolerance level. Differences of over 10% between simulated and measured values occur for five different systems; the capacity and efficiency for Manufacturer B's 10 SEER system, the capacity for Manufacturer B's 12 SEER system, the compressor power for Manufacturer D's 10 SEER and 12 SEER systems, and the capacity and efficiency for Manufacturer J's 10 SEER system. Obviously, MODCON's performance at the 17°F (-8.3°C) test condition does not match its performance at the cooling and 47°F (8.3°C) test conditions. But although over half of the simulated values exceed the 5% tolerance level, average differences between simulated and measured values are still well below 10%.

Simulation Modeling Changes

Based on manufacturer recommendations, changes were made to MODCON to try and improve its performance. In addition, guidelines were established for conducting simulation runs with the modified version of MODCON.

Because simulation errors at estimating efficiency seemed to be due more to problems with MODCON's ability to predict system capacity rather than predicting compressor power, manufacturers recommended including correction factors in MODCON to reduce the estimated system capacity. Correction factors (also known as capacity reduction factors) of 95% for the cooling test conditions, 94% for the 47°F (8.3°C) heating test condition, and 90% for the 17°F (-8.3°C) test condition were adopted and the appropriate changes were made to MODCON.

In addition to adopting the above correction factors for capacity, a new method was adopted for determining the refrigerant pressure drop in the suction line. The appropriate changes in MODCON were made to include the following calculation for the suction line pressure drop.

$$\Delta P \ = \ c \bullet m^{1.8} \tag{2}$$

where ΔP = suction line pressure drop (lb_f/in^2)
c = constant
m = refrigerant mass flow rate (lb_m/hr)

The constant in the above equation is established based on a reference suction line pressure drop and refrigerant mass flow rate.

In addition to the above changes that were made to MODCON, manufacturers recommended guidelines for which to conduct simulation runs. The guidelines pertained to acceptable values for refrigerant pressure drops in the heat exchanger coils and in the suction line. These guidelines were based on manufacturer design practices for achieving particular refrigerant pressures through out the system. Thus, although test data may indicate pressure drop values which differ from those designed for by manufacturers, simulations should be conducted to match the targeted rather than the measured pressure drop values. Table 2 provides the pressure drop guidelines.

Table 2: Refrigerant Pressure Drop Guidelines

Location	ΔP
Indoor Coil	
Cooling mode	20.7 kPa (3.0 lb_f/in^2)
Heating mode	determined from MODCON
Outdoor Coil	20.7 kPa (3.0 lb_f/in^2) to 34.5 kPa (5.0 lb_f/in^2)
Suction Line	
Cooling Capacity < 11.4 kW (39,000 Btu/hr)	31.0 kPa (4.5 lb_f/in^2) @ mass flow rate = 250 kg/hr (550 lb_m/hr)
Cooling Capacity ≥ 11.4 kW (39,000 Btu/hr)	31.0 kPa (4.5 lb_f/in^2) @ mass flow rate = 408 kg/hr (900 lb_m/hr)

Comparison of Simulation Results to Manufacturer Test Data after Modeling Changes

For the four sets of manufacturer data, new simulation runs were conducted to determine whether the modifications made to MODCON, along with the new guidelines for establishing refrigerant pressure drops, improved the performance of the model. As was done for the prior simulation runs, measured quantities from the manufacturer test data were used by MODCON to conduct the new simulation runs. These measured quantities were the same as those that were specified for the prior simulation runs. The only difference being that the new guidelines for establishing suction line and heat exchanger coil pressure drops were used instead of the measured test data. Also different from the prior simulations, refrigerant-side pressure drop multipliers (input variables to MODCON) were not used to regulate the simulated heat exchanger pressure drops. MODCON input variables which describe the number of parallel circuits in the heat exchangers were used instead.

Similar to Figures 1 through 4, Figures 5 through 8 depict the percentage differences between the test data and the simulation results for capacity, efficiency, and compressor power. But in the latter set of figures a "before" and "after" set of comparisons are provided. The "before" set of comparisons are identical to those in Figures 1 through 4. They depict the percentage differences between the test data and the simulation results prior to any capacity correction factors being incorporated into MODCON and any pressure drop guidelines being adopted for conducting simulation runs. The "after" set of comparisons depict the percentage differences after both the correction factors were added to MODCON and the new pressure drop guidelines were adhered to. By providing both sets of data, direct visual comparisons can be more easily made. As in Figures 1 through 4, Figures 5 through 8 are divided into two sections; the section on the left details the comparisons for the minimum efficiency model (nominal cooling efficiency of 10 SEER) while the section on the right details the comparisons for the high-efficiency model (nominal cooling efficiency of either 12 or 13 SEER). In each figure, labels are provided above each set of comparisons for capacity, efficiency, and compressor power designating the corresponding model and test condition (e.g., 10 SEER-95°F designates data comparisons for the 10 SEER model rated at the 95°F (35°C) test condition). The "before" and "after" labels appear beneath each set of comparisons designating which set were determined pre- and post MODCON modifications.

As depicted in Figures 5 through 8, the effect of following the new pressure drop guidelines and incorporating the capacity correction factors into MODCON improves the capability of the model to estimate heat pump performance. Starting with the heat pump's cooling side, as opposed to the simulations done prior to any MODCON modifications, there are now no occurrences of differences of over 10% between the simulated and measured test data for any of the systems. And on an average basis, there is also a reduction in the differences between simulated and measured capacity and efficiency due to the modeling changes. Where prior to MODCON modifications the average differences between simulated and measured values for capacity, efficiency, and compressor power for all four sets of manufacturer data were 5.8%, 6.7%, and 2.4%, respectively, now the average differences are 4.0%, 5.4%, and 2.6%, respectively.

With regard to the 47°F (8.3°C) heating test condition for heat pumps, the modeling changes slightly improved MODCON's ability to estimate capacity and compressor power. Where prior to MODCON modifications the average differences between simulated and measured values for capacity, efficiency, and compressor power for all four sets of manufacturer data were 6.7%, 5.2%, and 3.6%, respectively, now the average differences are 6.3%, 5.5%, and 2.0%, respectively. The modified version of MODCON actually does a poorer job of estimating efficiency on an average basis, but the degradation in performance is too small to be considered significant. There were no changes in the number of systems exhibiting simulation errors beyond the 5% tolerance level. Prior to modeling changes the following five systems exhibited simulation errors greater than ±5%; Manufacturer B's 10 SEER and 12 SEER systems, Manufacturer D's 12 SEER system, and Manufacturer J's 10 and 12 SEER systems. With the modeling changes Manufacturer B's 10 SEER and 12 SEER systems were replaced by Manufacturer A's 10 SEER and 12 SEER systems as those systems exhibiting simulation errors beyond ±5%.

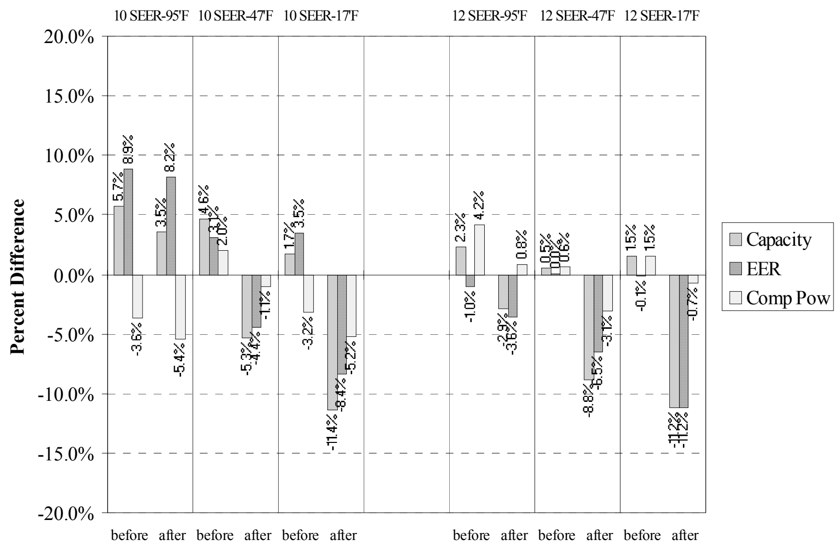

Figure 5: Manufacturer A: Test Data vs.Simulation Results; comparisons before and after simulation modeling changes

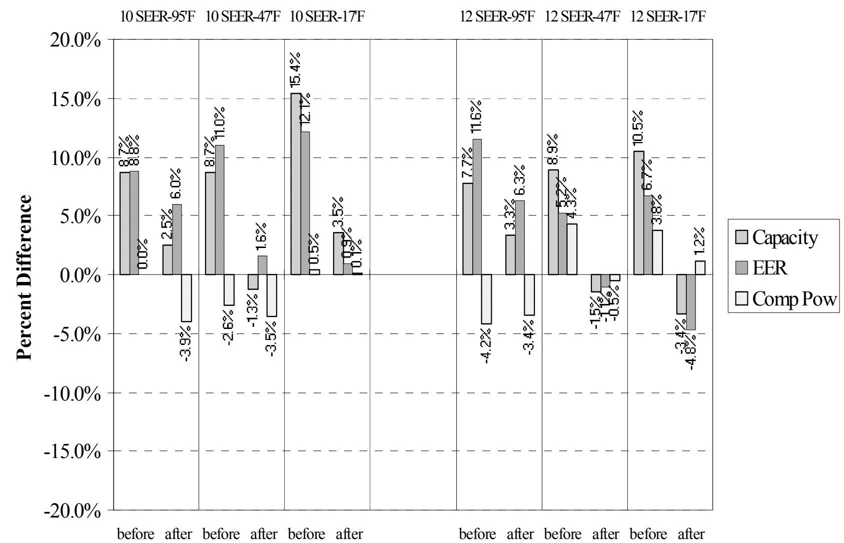

Figure 6: Manufacturer B: Test Data vs.Simulation Results; comparisons before and after simulation modeling changes

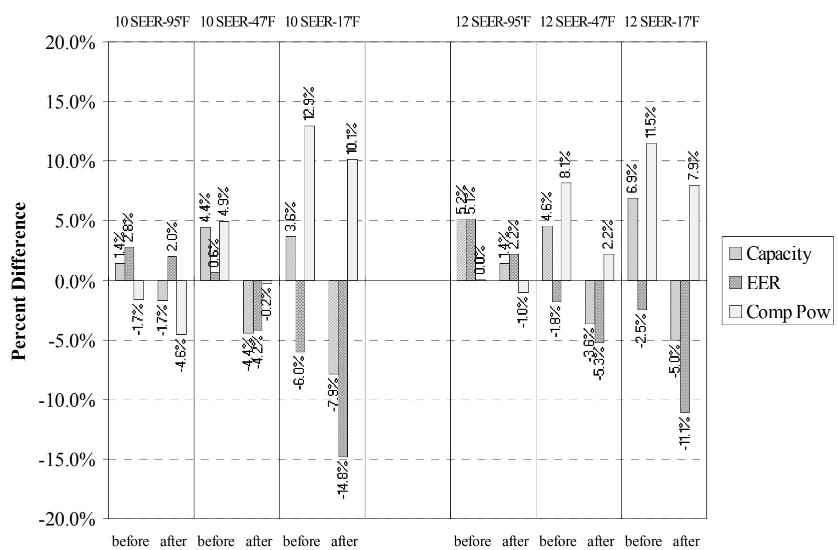

Figure 7: Manufacturer D: Test Data vs. Simulation Results; comparisons before and after simulation modeling changes

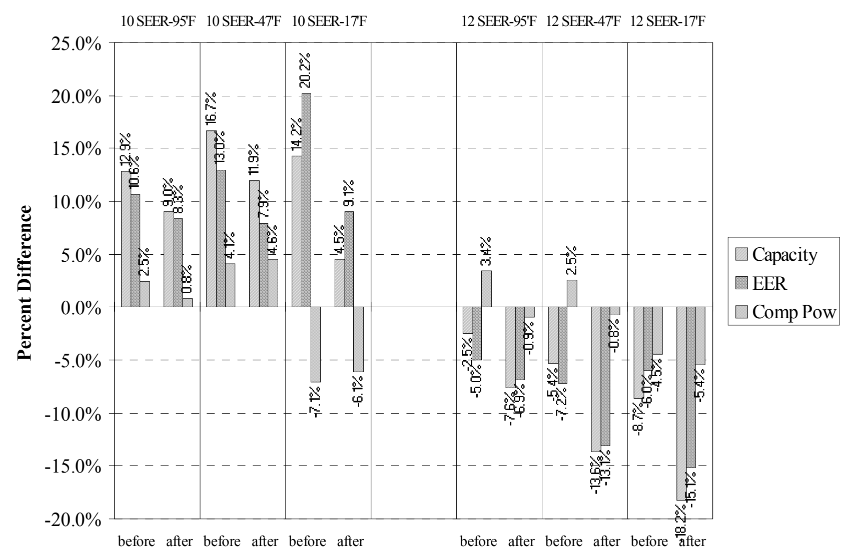

Figure 8: Manufacturer J: Test Data vs. Simulation Results; comparisons before and after simulation modeling changes

At the 17°F (-8.3°C) test condition, the simulation modeling changes actually worsen the performance of MODCON. Prior to simulation modeling changes, average differences between simulated and measured values for capacity, efficiency, and compressor power for all four sets of manufacturer data were 7.8%, 7.1%, and 5.6%, respectively. With the modeling changes the average differences are now 8.1%, 9.4%, and 4.6%, respectively. Only compressor power is better estimated with the modified version of MODCON. In addition, of the 24 comparisons depicted between simulated and measured values in Figures 5 through 8, 63% (or fifteen simulated quantities) are beyond the 5% tolerance level. This is in comparison to 54% (or thirteen simulated quantities) which were beyond the 5% tolerance level prior to any modeling changes were made.

New Correction Factors for Heating Test Conditions

Because of MODCON's relatively poor performance at the heating test conditions, correction factors other than 94% and 90% for the 47°F (8.3°C) and 17°F (-8.3°C) test conditions, respectively, were tested in an attempt to reduce the differences between simulated results and the test data. It was determined that a correction factor of 96% yielded the best results for both the 47°F (8.3°C) and 17°F (-8.3°C) test conditions. Figures 9 through 12 depict the percentage differences between the test data and the simulation results for capacity, efficiency, and compressor power based on the use of new (96%) and old (94% and 90%) correction factors. Figures 9 through 12 are divided into two sections; the section on the left details the comparisons for the minimum efficiency model (nominal cooling efficiency of 10 SEER) while the section on the right details the comparisons for the high-efficiency model (nominal cooling efficiency of either 12 or 13 SEER). In each figure, labels are provided above each set of comparisons for capacity, efficiency, and compressor power designating the corresponding model and test condition (e.g., 10 SEER-47°F designates data comparisons for the 10 SEER model rated at the 47°F (8.3°C) test condition). Labels appear beneath each set of comparisons designating whether the simulation results were determined with or without correction factors. A label of "before" designates simulation results determined without correction factors. Simulation results based on old correction factors are designated with labels of either "CF=0.94" (for the 47°F (8.3°C) test condition) or "CF=0.90" (for the 17°F (-8.3°C) test condition). Simulation results based on new correction factors are designated with labels of "CF=0.96".

As depicted in Figures 9 through 12, the effect of incorporating the new capacity correction factors into MODCON improves the capability of the model to estimate heat pump heating performance. Starting with the 47°F (8.3°C) heating test condition, the new correction factor of 96% yields average differences between simulated and measured values of 5.2%, 4.8%, and 2.0% for capacity, efficiency, and compressor power, respectively, for all four sets of manufacturer data. This is in contrast to average differences of 6.7%, 5.2%, and 3.6% for the case prior to MODCON modifications and 6.3%, 5.5%, and 2.0% for the case where a correction factor of 94% was used. With the use of the 96% correction factor, only the average difference for capacity exceeds the 5% tolerance level, but only by an amount of 0.2 percentage points.

At the 17°F (-8.3°C) test condition, the new correction factor of 96% yields average differences between simulated and measured values of 6.4%, 7.1%, and 4.6% for capacity, efficiency, and compressor power, respectively, for all four sets of manufacturer data. This is in contrast to average differences of 7.8%, 7.1%, and 5.6% for the case prior to MODCON modifications and 8.1%, 9.4%, and 4.6% for the case where a correction factor of 90% was used. On an average basis, the 96% correction factor yields significantly better results than the 90% correction factor, although the average differences for capacity and efficiency still exceed the 5% tolerance level by 1.4 and 2.1 percentage points, respectively.

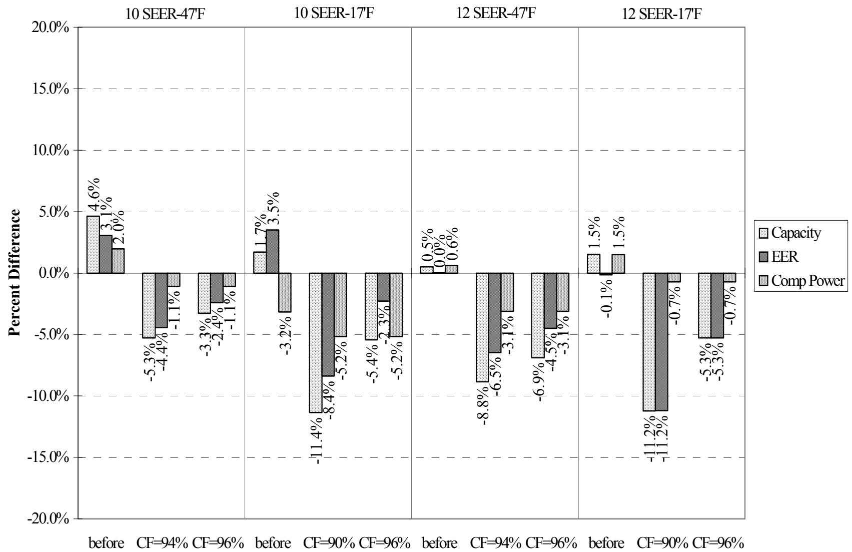

Figure 9: Manufacturer A: Test Data vs. Simulation Results; heating test condition comparisons with and without correction factors

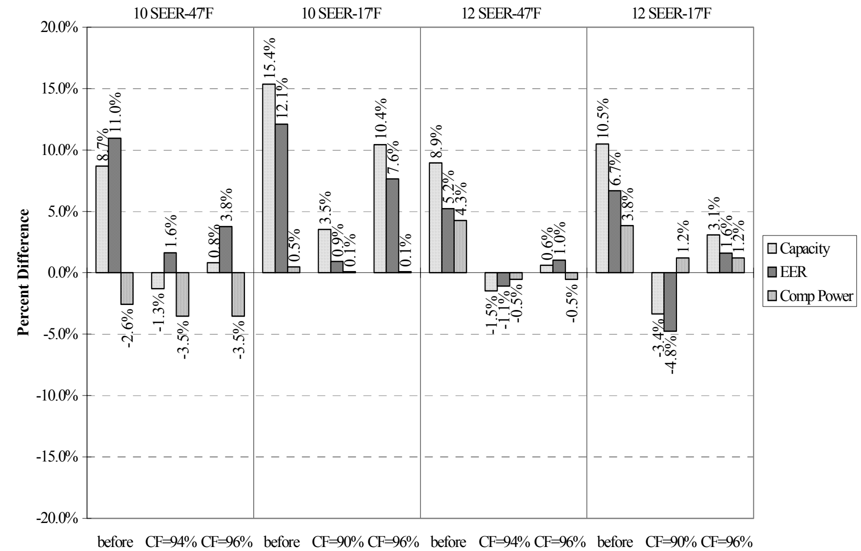

Figure 10: Manufacturer B: Test Data vs. Simulation Results; heating test condition comparisons with and without correction factors

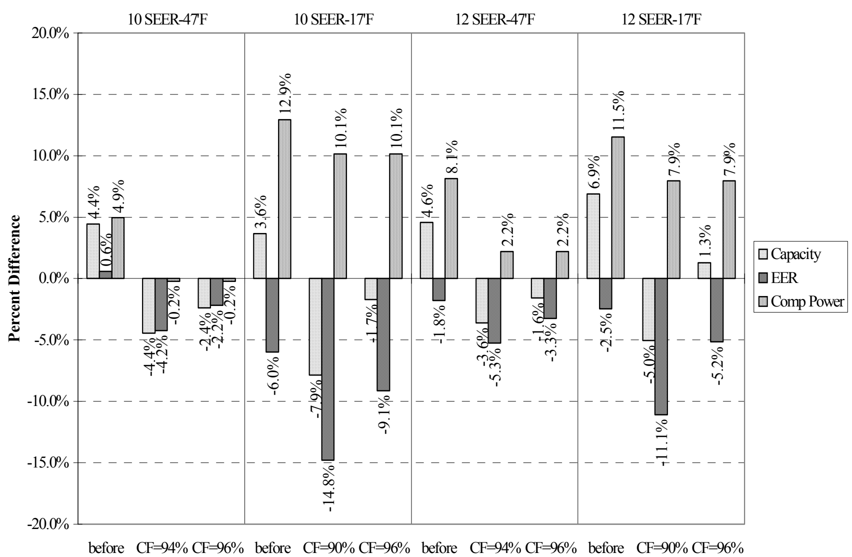

Figure 11: Manufacturer D: Test Data vs. Simulation Results; heating test condition comparisons with and without correction factors

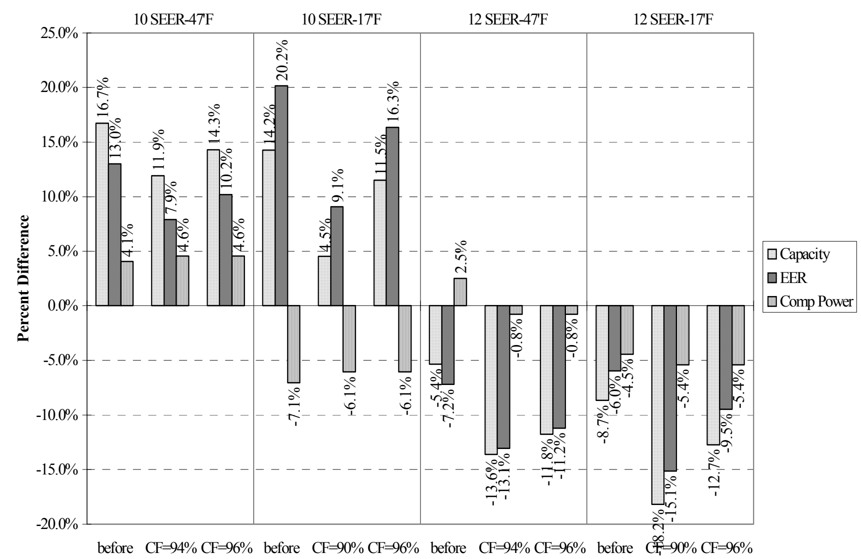

Figure 12: Manufacturer J: Test Data vs. Simulation Results; heating test condition comparisons with and without correction factors

Additional Guidelines

As a result of conducting the validation runs with the manufacturer test data, it became evident how the air flow rate over the indoor and outdoor heat exchanger coils, the power consumption of the fan motors, the compressor shell heat loss rate, and the heat gains and losses in the refrigerant lines varied from test condition-to-test condition. It should be emphasized that these derived relationships do not necessarily represent the behavior of any single heat pump unit. Rather, the relationships represent the average behavior of all the heat pump equipment which were analyzed. It should also be noted that the 82°F (27.8°C) test condition is defined as the design test condition. As a result, several of the relationships are defined in reference to it. The 82°F (27.8°C) test condition is known as the design test condition because the SEER for single-speed heat pumps is a function of only two quantities; the EER at the 82°F test condition and the degradation coefficient (C_D). Thus, if manufacturers want to maximize the SEER of a single-speed system, they will do so by designing for the greatest EER at the 82°F (27.8°C) test condition.

Manufacturer test data indicated that the air flow rate over each coil stayed relatively constant regardless of the heat pump test condition. That is, if the air flow rates delivered across the indoor and outdoor coils during the 82°F (27.8°C) test condition were 0.57 and 1.18 m^3/s (1200 and 2500ft^3/min), roughly the same air flow rates would also be delivered at the 95°F (35°C) test condition as well as both the 47°F (8.3°C) and 17°F (-8.3°C) test conditions.

With regard to fan motor power consumption, test data indicated that the indoor fan motor consumed roughly the same amount of power during all steady-state test conditions. For the outdoor fan motor, power consumption varied amongst the four steady-state cooling and heating test conditions. From the manufacturer test data, general relationships were derived as a function of the power consumed at the 82°F (27.8°C) test condition. The relationships are provided below.

$$\begin{aligned} OFMP_{95°F} &= 0.9875 \bullet OFMP_{82°F} \\ OFMP_{47°F} &= 1.0792 \bullet OFMP_{82°F} \\ OFMP_{17°F} &= 1.1333 \bullet OFMP_{82°F} \end{aligned} \tag{3}$$

where $OFMP_{82°F}$ = outdoor fan motor power consumption @ 82°F (27.8°C) test condition
$OFMP_{95°F}$ = outdoor fan motor power consumption @ 95°F (35°C) test condition
$OFMP_{47°F}$ = outdoor fan motor power consumption @ 47°F (8.3°C) test condition
$OFMP_{17°F}$ = outdoor fan motor power consumption @ 17°F (-8.3°C) test condition

For purposes of simulation modeling, the compressor shell heat loss rate was defined as a fraction of the compressor input power. For air conditioners and heat pumps operating in the cooling mode, the test data indicated that the compressor shell heat loss rate was approximately 9% of the compressor input power. For the heat pumps operating in the heating mode, test data indicated the heat loss rate as being 22.5% of the compressor input power.

With regard to the refrigerant line heat gains and losses, Table 3 provides the representative values as indicated by the manufacturer test data.

Table 3: Refrigerant Line Heat Gains and Losses

	Suction Line Heat Gain		Discharge Line Heat Loss		Liquid Line Heat Loss	
Test Condition	*Watt*	*Btu/hr*	*Watt*	*Btu/hr*	*Watt*	*Btu/hr*
82°F (27.8°C) test condition	293	1000	205	700	205	700
95°F (35°C) test condition	293	1000	205	700	205	700
47°F (8.3°C) test condition	2.93	10	205	700	205	700
17°F (-8.3°C) test condition	2.93	10	205	700	205	700

The above relationships are instrumental to conducting simulation modeling when designing a single-speed system with the simulation model. With the above relationships, once a system's behavior is defined at the 82°F (27.8°C) test condition (i.e., the design test condition), the air flow rates and the power consumption of each fan motor are known for all other steady-state test conditions.

In addition to the above relationships, guidelines were also established for the modeling of flow control devices. It was learned from the validation runs that MODCON's submodel for thermostatic expansion valves (TXV) did not perform adequately. Thus, an alternative method for modeling both TXVs and short tube orifices was developed based on manufacturer recommendations. This method laid a framework for simulating the performance of a single-speed heat pump system. The framework relied on MODCON's capability of determining the required refrigerant charge of a system. For the cooling side of heat pumps, the framework consists of first optimizing the performance of the equipment at the 82°F (27.8°C) test condition (i.e., the design test condition) and then calculating the performance at the 95°F (35°C) test condition using key variables determined from the 82°F (27.8°C) simulation run. For the heating side of heat pumps, the performance at the 47°F (8.3°C) test condition is optimized (independent of what was done on the cooling side) and then the performance at the 17°F (-8.3°C) test condition is calculated using key variables determined from the 47°F (8.3°C) simulation run. For single-speed systems with short tube orifices, the following steps were developed for conducting the simulation runs.

1. Optimize cooling performance at 82°F (27.8°C) test condition
   a. Fix the amount of superheat at 20°F (11.1°C)
   b. Vary the amount of subcooling in the range of 10°F (5.6°C) to 20°F (11.1°C) to optimize efficiency
2. Calculate cooling performance at 95°F (35°C) test condition
   a. Specify same orifice size as that determined from the 82°F (27.8°C) simulation run
   b. Specify same refrigerant charge as that determined from the 82°F (27.8°C) simulation run
3. Optimize cooling performance at 47°F (8.3°C) test condition (independent of cooling side)
   a. Fix the amount of superheat at 0°F (0°C)
   b. Vary the amount of subcooling in the range of 15°F (8.3°C) to 30°F (16.7°C) to optimize efficiency
4. Calculate heating performance at 17°F (-8.3°C) test condition
   a. Specify same orifice size as that determined from the 47°F (8.3°C) simulation run
   b. Specify same refrigerant charge as that determined from the 47°F (8.3°C) simulation run
5. Add accumulator to system (if one does not already exist)
6. Repeat steps 1 through 4 if accumulator needs to be added to system

For single-speed systems with TXVs, the following steps were developed for conducting the simulation runs.

1. Optimize cooling performance at 82°F (27.8°C) test condition

   a. Fix the amount of superheat at 20°F (11.1°C)
   b. Vary the amount of subcooling in the range of 10°F (5.6°C) to 20°F (11.1°C) to optimize efficiency
2. Calculate cooling performance at 95°F (35°C) test condition
   a. Specify same superheat as that specified for the 82°F (27.8°C) simulation run
   b. Specify same refrigerant charge as that determined from the 82°F (27.8°C) simulation run
3. Optimize cooling performance at 47°F (8.3°C) test condition (independent of cooling side)
   a. Fix the amount of superheat at 7.5°F (4.2°C)
   b. Vary the amount of subcooling in the range of 10°F (5.6°C) to 20°F (11.1°C) to optimize efficiency
4. Calculate heating performance at 17°F (-8.3°C) test condition
   a. Specify same superheat as that specified for the 47°F (8.3°C) simulation run
   b. Specify same refrigerant charge as that determined from the 47°F (8.3°C) simulation run
5. Add accumulator to system only if compressor charge limit has been exceeded
6. Repeat steps 1 through 4 if accumulator needs to be added to system

As with the guidelines for establishing air flow rates and fan motor power consumption, the above steps for conducting simulation runs are instrumental in the design of a single-speed heat pump using the simulation model. The frameworks arising from the need to model flow control devices provide a structured method in which to simulate the performance of a heat pump system.

CONCLUSIONS

A modified version of the Oak Ridge National Laboratory Heat Pump Design Tool is able to predict (on an average basis) the capacity, efficiency, and compressor power of an air-source heat pump to within 5% of measured test data at the DOE "A" cooling test condition (outdoor ambient air temperature of 95°F (35°C)) and the high-temperature heating test condition (outdoor ambient air temperature of 47°F (8.3°C)). Simulation results are not within 5% of the measured test data at the low-temperature heating test condition (outdoor ambient air temperature of 17°F (-8.3°C)) although modifications do yield simulation results which are on average better than those before modifications were made to the simulation model.

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